AFIT/GAE/ENY/95D-19

THESIS

INFLUENCE OF A MOVING ENDWALL ON THE TIP CLEARANCE VORTEX IN AN AXIAL COMPRESSOR CASCADE

Lawrence J. Peter
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THESIS

Presented to the Faculty of the Graduate School of Engineering

of the Air Force Institute of Technology

Air University

In Partial Fulfillment of the

Requirements for the Degree of

Master of Science in Aeronautical Engineering

Lawrence J. Peter

Major, USAF

December 1995

Approved for public release; distribution unlimited

Acknowledgments

This thesis is the final result of a large amount of time, thought, and effort. I would never have finished this investigation without the assistance and ideas of a great many people. I would like to take this opportunity to thank these people for their aid and support.

I would like to thank my advisor Dr. Paul I. King for his help in directing my efforts during this project. His continual questions concerning procedures and assumptions prevented my jumping to premature conclusions, and helped to pull me out of several blind alleys.

This investigation would never have been completed were it not for the craftsmanship, ideas and efforts of the AFIT Model Shop. Mr. Jack Tiffany and his team exceeded all my expectations with their skill and attention to detail.. Mr. Timothy Hancock and Mr. Jan DeValley were invaluable in the conceptualization and design of the moving wall modification which they then constructed in its entirety. Mr. David Driscoll's efforts were critical in the manufacture of the new blading and in fine tuning the rotor drive components.

The AFIT laboratory technicians Mr. Jay Anderson, Mr. Andrew Pitts, Mr. Charles McNeely, Mr. Daniel Rioux and Mr. Mark Deriso were very helpful. Their assistance in everything from electronics to plumbing allowed me to get and keep the equipment operating. Mr. Andrew Pitts efforts in troubleshooting the traverse were greatly appreciated.

I would also like to thank Major Don Thompson for his help in gathering research materials and his useful suggestions for this report.

Finally, and most importantly, I would like to thank my wife Maria for her love, support, understanding and encouragement throughout this project. Her patience and efforts to minimize distractions were essential to my finishing this research. In addition, I would like to thank my daughter Amanda for helping me keep things in perspective and for understanding when daddy couldn't take time to play because I had to "study".

Lawrence J. Peter

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Symbol

AFIT	Air Force Institute of Technology
AR	Aspect ratio
PS	pressure surface
RPM	revolutions per minute
SS	suction surface
C_{DA}	coefficient of annulus drag
C_{DS}	coefficient of drag for secondary effects
C_{DP}	coefficient of profile drag
C_D	coefficient of drag
C_L	coefficient of lift
C_p	specific heat at constant pressure (J/kg-^{o}K)
Cp	Static pressure rise coefficient
C_x	flow through velocity (meters/sec)
L	characteristic length (meters)
R	specific gas constant (J/kg-^{o}K)
T_{static}	static temperature (^{o}K)
T_{total}	total temperature (^{o}K)
U	wall speed relative to blade tip (m/sec)
V_1	throat flow velocity (m/sec)

V_2	exit flow velocity (m/sec)
V_{local}	local flow velocity at a given point (m/sec)
X, x	axial coordinate
Y, y	pitchwise coordinate
Z, z	spanwise coordinate
a	point of maximum camber measured from leading edge (cm)
c	blade chord (cm)
h	blade height or span (cm)
s	one blade spacing (cm)
t	tip clearance gap (cm)
i	incidence angle (degrees)
i^*	nominal incidence angle (degrees)
$\dot{m}_{center}$	total mass flow for the center region (kg/sec)
$\dot{m}_{region}$	total mass flow for a given flow region (kg/sec)
$\dot{m}_{root}$	mass flow for the root region (kg/sec)
$\dot{m}_{tip}$	mass flow for the tip region (kg/sec)
$\dot{m}_{window}$	mass flow for the testing window (kg/sec)
p_{static}	static pressure (Pa)
p_1	static pressure at throat upstream of cascade (Pa)
p_2	static pressure downstream of cascade (Pa)
P_{total}	total or stagnation pressure (Pa)

P_{rake}	total pressure sensed by total pressure rake (Pa)
P_{loss}	loss of total pressure due to passage losses (Pa)
P_{local}	total pressure at a local point (Pa)
P_{tank}	pressure measured at stilling chamber (Pa)
Re	Reynolds number
α	Coefficient of thermal expansion (cm/cm-^{o}K)
α_m	mean air angle (degrees)
α_1	air inlet angle (degrees)
α'_1	blade inlet angle (degrees)
α_2	air outlet angle (degrees)
α'_2	blade outlet angle (degrees)
ε	deflection (degrees)
ϕ	flow coefficient (C_x/U)
ϕ^{-1}	inverse of flow coefficient (U/C_x)
ζ	stagger angle (degrees)
δ	strain growth (cm/cm)
δ_T	thermal growth (cm/cm)
γ	ratio of specific heats
ρ	density (kg/m^3)
θ	blade camber angle (degrees)
ϖ_{root}	mass averaged total pressure loss coefficient for root region
ϖ_{tip}	mass averaged total pressure loss coefficient for tip region

ϖ_{center} mass averaged total pressure loss coefficient for center region

ϖ_{region} mass averaged total pressure loss coefficient for a given region

$\varpi_{tip\text{-}region}$ mass averaged total pressure loss coefficient for tip region averaged by tip region mass flow

ϖ_{ref} mass averaged total pressure loss coefficient for a reference tip clearance

$\varpi_{passage}$ mass averaged total pressure loss coefficient for the passage

ω_{local} total pressure loss coefficient at a local data point

ΔA incremental area surrounding representing a data point (meters^2)

Δy incremental pitchwise distance (meters)

Δz incremental spanwise distance (meters)

Abstract

This experiment involved the design, construction, validation and testing of a new facility for the investigation of vortices generated by compressor rotor blade tip clearance with a moving endwall. A five-tube pressure rake placed downstream of the trailing edge of a cascade of blades measured the pressure field for flow coefficients ranging from 20 to 1.66 and tip clearances of 0.33, 1.0, 1.7 and 2.4 percent chord. Contour plots of mass averaged pressure loss coefficient appear to show the no-flow tip vortex becomes entrained and diffused by the moving wall boundary layer. The high loss region near the moving wall contracts toward and extends toward the pressure side of the adjacent blade. This contraction results in a reduction in overall blockage in the passage with a corresponding reduction in passage losses, toward an apparent steady-state value, for increasing end wall speed and decreasing tip clearance.

INFLUENCE OF A MOVING ENDWALL ON THE TIP CLEARANCE VORTEX IN AN AXIAL COMPRESSOR CASCADE

I. Introduction

Background

Linear cascade wind tunnels are an important tool in the search for improving performance of aircraft turbine engines and understanding the flow field and loss mechanisms associated with turbomachinery blading. The ability to test compressor blade designs without the necessity of installing them in actual engines saves significant cost and time in research.

The interaction of the engine annulus boundary layer with the leakage flow across the blade tip gap is one of the major loss mechanisms in turbomachinery. Understanding this and other three-dimensional interactions is the continuing object of research on blade design and modeling in an effort to improve engine efficiency and performance.

Summary of Previous Research

The blade tip region has long been recognized as an important area of study due to the losses associated with the annulus boundary layer interacting with the tip leakage flow and spanwise flows. Herzig et al.(1954) conducted several experiments using a moving belt to simulate the engine annulus wall moving past the blade tips. This research showed

the existence of a so-called scraping vortex created by the wall boundary layer being scraped off by the pressure side of the blade tip moving relative to the annulus. Since Herzig's work, numerous studies of the endwall region of compressor cascades have been performed using linear cascades with stationary endwalls to identify and investigate the various flow structures in a compressor rotor passage. Kang and Hirsch, (1991, 1993).

Several studies were conducted at Pennsylvania State University on a rotating compressor rig with probes designed to follow a single blade as it travels past the annulus wall: Sitram and Lakshminarayana, (1983), Lakshminarayana, Sitram and Zhang, (1986), Lakshminarayana, Zaccaria, and Marathe, (1995). Specialized rigs of this type have permitted the simultaneous study of multiple flow interactions. Findings of these investigations, however, include the observation that stationary cascade data does not generally prove a good predictor of passage losses for turbomachinery. Typical cascades for example, are unable to model the skewing of the inlet flow caused by interaction with the annulus boundary layer, (Cumpsty, 1989:334).

Endwall research has allowed some categorization of the different flow structures associated with the tip region; however, the combined interaction of these flow effects is extremely difficult to study in actual engines.

Objective

The objective of this thesis was to design, construct, verify and test a modification to the AFIT linear compressor cascade facility. This modification was intended to permit the study of several three-dimensional flow effects and in particular the interactions of blade tip clearance and the annulus wall boundary layer. This was accomplished by

replacing a portion of the front wall of a linear cascade with the edge of a large rotor. The rotor diameter was sufficiently large to appear approximately flat for small distances along the cascade. Spinning this large rotor so that its edge reaches a large percentage of the flow velocity was intended to simulate the relative motion of an engine annulus next to the blade tips. While unable to model the radial effects of rotation, this modification should permit the study of inlet flow skewing, tip clearance, and annulus boundary layer interactions.

Once constructed, several tip clearances from 0.33% to 2.4% chord (1 percent to 3 percent chord are typically seen in aeronautical applications) were tested at a range of flow coefficients (axial through-flow velocity/blade speed) from 20 to 1.66. While this apparatus partially models the flow influences of a rotor blade at the tip, it also models an unshrouded compressor stator. The compressor stator has the blade fixed at the outer annulus wall, and its tip, with clearance, located at the hub. While a compressor rotor tip will operate with flow coefficient of approximately 0.5, the flow coefficient for the stator tip would be on the order of approximately 10.0 due to the smaller radius of the hub, a value in the region of those tested in this project. The Pennsylvania State University compressor rig mentioned earlier (Lakshminarayana et al.,1986) uses a flow coefficient of approximately 0.56 for the tip, however, relative to the hub, the flow coefficient is approximately 7.0, a value also in the range modeled by the slow speed region of this experiment.

While values of ϕ typically range near 0.5 for a compressor rotor at its design point, the engine must go through the higher range of ϕ's during start-up and idle. Though

optimizing an engine for high mass flows and pressure ratios is required, knowledge of the off-design case of low RPM and low mass flow is also required. This flow region is well modeled by this experiment for both the rotor and stator tip regions.

Stagnation pressure loss contour plots were constructed to visualize the endwall flow field and the penetration of three-dimensional effects into the passage. Pressure losses were then quantified to determine a relationship with tip clearance and relative wall motion. From these relationships, an empirical relation was developed to predict passage losses for a wide range of tip clearances. Finally, this new capability was applied to the performance of crenulated blades.

A crenulated blade design proposed by Spacy (1993) was also evaluated with tip clearance and moving wall effects. Earlier studies of blade crenulations in linear cascades have shown evidence of improvements in wake mixing and total pressure losses. This modification will test a promising crenulation design with tip clearance and moving wall effects to determine if the cascade results remain valid.

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II. Theory

Three Dimensional Flows in the Endwall Region

The tip region of a compressor rotor is the site of strong three-dimensional interactions that are not easily modeled with numerical or analytical techniques. Much cascade and engine research has concentrated on this area to gain an understanding of the physics of the flowfield within a compressor.

A major limitation of cascade facilities lies in their inability to simulate many of the three-dimensional effects inherent in turbomachinery. Several studies have explored individual flow phenomena which can be observed. One must be cautious, however, in attempting to isolate single aspects of endwall flow, (Sitram and Lakshminarayana, 1983). Individual effects cannot always be isolated due to their compounding and synergistic effects. In addition three-dimensional effects seen in cascade research do not necessarily apply to the different conditions present in an engine, (Lakshminarayana et al.,1986:30), (Cumpsty, 1989:334). For this reason, linear cascade facilities are typically used to model two-dimensional flows, with actual compressors providing the laboratory for three-dimensional studies. The restriction of linear cascade facilities to two-dimensional flows limits their usefulness in investigating blade-tip clearance effects, as well as boundary layer interactions that occur between the engine annulus wall and spinning compressor or turbine blades. As will be shown in this study, however, the two-dimensional limitation of cascades can be overcome by proper modeling and design.

Figure 1 from Cumpsty (1989:332) shows the various boundary layer and flow interactions present in the passage endwall region. They consist of the annulus wall boundary layer, the blade boundary layer, tip leakage flow, secondary flow, radial flow and the scraping vortex. These are the principal sources of three-dimensional flow effects in the endwall region within a compressor. This experiment dealt primarily with the tip leakage flow and its interaction with the annulus boundary layer.

Secondary flow has several different connotations, depending upon the writer involved, and generally refers to the group of flow effects associated with the end wall region, including tip clearance and viscous effects. Due to the ambiguity of definition, the individual flow phenomena will be discussed individually where appropriate.

The annulus boundary layer is the viscous shear layer developed by the axial flow through the engine with the no-slip velocity condition occurring on the engine annulus. The thickness of the wall boundary layer will be a function of the flow Reynolds number and the distance it has had to develop as well as the pressure gradients involved. If a multiple stage compressor is used, the boundary layer can be relatively large due to its continued growth throughout the engine, (Cumptsy, 1989:80). The thickness of the wall boundary layer will also depend upon whether the flow is laminar or turbulent.

The rotor blade boundary layer is created by the flow over the blades. On the pressure surface it will generally be small due to the favorable pressure gradient, while on the suction side it will be a function of the blade loading and may be relatively thick in highly loaded blades where separation has begun to occur. Losses associated with the

blade boundary layer are generally referred to as profile losses and are relatively easy to observe in the blade wake.

Tip leakage flow results from the pressure gradient between the pressure and suction sides of the blades and the necessity for a clearance gap to prevent contact between the rotor blades and the annulus wall. The presence of a blade tip clearance alters the blade pressure distribution and creates a tip vortex whose strength depends upon the amount of clearance. This vortex removes the corner separation evident in a typical cascade while introducing additional losses and decreasing the pressure rise capability, (Cumpsty, 1989:355). Tip clearance, therefore, is of major importance in overall passage losses. While the annulus boundary layer interacts with the tip leakage flow, Storer and Cumpsty (1991:258) have used Navier-Stokes solvers to show that tip leakage is essentially an inviscid phenomena driven primarily by the pressure differences between the pressure and suction sides of the blade.

An important phenomenon occurring in this region is the circumferential flow component induced by the wall boundary layer on the axial flow of the engine, (Storer and Cumpsty, 1991:258). This skewing of the velocity component due to the boundary layer velocity gradient causes the blade tip within the boundary layer to see a different incidence than regions outside the boundary layer. Typically, cascade wind tunnels exhibit what Cumpsty refers to as collateral flow in which the entire blade span sees the same inlet velocity and incidence regardless of span location and endwall boundary layer thickness. Cumpsty (1989:334) has concluded, therefore, that typical compressor cascade data does

not necessarily model engine flows due to this inability to simulate the skewing of the incoming velocity vector by the annulus boundary layer interaction.

Methods of Studying EndWall Effects

To study the endwall flow regime, one must include the primary flow phenomena in order to have a model applicable to actual turbomachinery. Ideally, this would be done with an actual compressor. Unfortunately, attempting to study the entire range of details in moving blades is a problem which requires specialized equipment such as present in the rotating compressor rig at Pennsylvania State University.

If an engine's radius is large enough, the annulus can be approximated as a flat wall. A linear cascade essentially models an annulus of infinite radius of curvature. In an engine, however, the blade tip is moving relative to the annulus while linear cascades typically have fixed walls at the blade tip. If the linear cascade tip wall were moved rapidly past the blade tips, it should simulate the motion of the blades past the engine annulus wall. This approach has been applied in several compressor and turbine studies with the front wall of a linear cascade replaced by a moving belt to study the moving wall effects, (Herzig, 1954), (Yaras et al., 1992). A limitation with belts to simulate a moving wall lies in the wall speeds achievable before encountering problems with belt tracking and vibration. For accurate simulations, the belt speed must be of close order of the flow velocity through the cascade. The flow velocities associated with belt moving wall simulations are limited to those attainable by the belt, which are not necessarily representative of those seen in actual compressor operation.

The purpose of this study then was to design and construct a system capable of realistically simulating a moving wall to observe the various flow effects at the blade tip and within the passage of a compressor rotor stage, while taking advantage of precise parameter control and accuracy of measurement afforded by a linear cascade.

<u>The Total Pressure Loss Coefficient for Quantifying Endwall Losses</u>

A common means for comparing passage losses is the mass averaged total pressure loss coefficient, ϖ . This quantity can be calculated for individual points or for the entire passage between two blades. For each data point in a passage, the value of ϖ and the local value of mass flow are computed:

$$\varpi_{local} = \frac{\Delta P_{local}}{\frac{1}{2} \cdot \rho \cdot V_1^2} \tag{1}$$

where the numerator is found by subtracting the total pressure measured at a local point from the total pressure of the flow P_{total} upstream of the blade entry plane.

$$\Delta P_{local} = P_{total} - P_{local} \tag{2}$$

P_{local} is measured at the point of interest with a total pressure rake or pitot tube.

To compute the local mass flow, several parameters must first be determined. Density, ρ, is computed from the perfect gas law

$$p_{static} = \rho \cdot R \cdot T_{static} \tag{3}$$

with static pressure, p_{static}, measured using static pressure ports located on the tunnel wall at the x location of the pressure rake. Static temperature, T_{static}, is computed from the known value of total pressure at a point through the isentropic relations:

$$\frac{p_{static}}{P_{local}} = \left(\frac{T_{static}}{T_{total}} \right)^{\frac{\gamma}{\gamma - 1}} \tag{4}$$

which can be rewritten as:

$$T_{static} = T_{total} \cdot \left(\frac{p_{static}}{P_{local}} \right)^{\frac{\gamma - 1}{\gamma}} \tag{5}$$

where γ is the ratio of specific heats, taken to be 1.4 for air in all calculations.

Velocity at each point is computed from the definition of total or stagnation temperature and the specific heat at constant pressure, C_p.

$$T_{total} = T_{static} + \frac{V_{local}{}^2}{2 \cdot C_p} \tag{6}$$

Solving Equation (6) for the local velocity,

$$V_{local} = \sqrt{2 \cdot \frac{\gamma \cdot R}{\gamma - 1} (T_{total} - T_{static})} \tag{7}$$

C_p for a calorically perfect gas can be expressed in terms of the specific gas constant, R, and the ratio of specific heats, γ.

Using these values, the local mass flow at each point is computed as

$$\dot{m}_{local} = \rho \cdot V_{local} \cdot \Delta A \qquad (8)$$

where ΔA represents the incremental area associated with a particular data point and equals the spanwise data spacing multiplied by the pitchwise step size, or $\Delta A = \Delta z \Delta y$

Finally, the mass averaged pressure loss coefficient for the passage can be computed as follows:

$$\varpi_{passage} = \frac{\sum_{i,j} (\varpi_{local} \cdot \dot{m}_{local})}{\sum_{i,j} \dot{m}_{local}} \qquad (9)$$

Note that this equation is normalized by the total mass flow for the passage with i representing steps in y and j steps in z.

The mass averaged pressure loss coefficient was selected as a primary quantity for comparison due to its inherent insensitivity to changes in mass flow.

<u>Empirical Estimates of Passage Losses</u>

Due to the difficulties in arriving at analytical means of predicting endwall losses, empirical relations have been developed to aid in design and analysis of engines. These passage losses can be computed based on empirical relations for secondary losses as

contained in Cumpsty, (1989) and Cohen et al.(1987). The blade geometry including spacing and gas angles are required to apply these equations.

Given the blade values, the losses for the passage are computed from the various components of drag. With

C_{DA} = annulus drag coefficient

C_{DS} = secondary drag coefficient

C_{DP} = parasite or profile drag coefficient,

these values result in the total drag coefficient for the span C_D,

$$C_D = C_{DP} + C_{DA} + C_{DS} \tag{10}$$

An empirical relation for the annulus drag coefficient is from Cohen et al. (1987):

$$C_{DA} = 0.02 \cdot \left(\frac{s}{h}\right) \tag{11}$$

with s/h the ratio of blade pitch to blade height.

In order to arrive at the secondary loss relation, the lift coefficient for the blade as well as the mean gas angle α_m must be computed. From Cohen et al. (1987),

$$\alpha_m = \tan^{-1}\left[\frac{1}{2} \cdot \left(\tan\alpha_1 + \tan\alpha_2\right)\right] \tag{12}$$

α_1 represents the blade air inlet angle, and α_2 the air outlet angle for the cascade as shown in Figure 2.

The profile drag is given by Howell's off-design compressor cascade performance data presented in Dixon, (1986:78) which applies to 2-dimensional cascades From Howell's Generalized Performance curves, (Dixon,1986:78), an initial value can be arrived at for C_{DP} based upon the incidence angle of the cascade and the nominal deflection.

The lift coefficient from Cohen et al. (1987:197) is given from the relation

$$C_L = 2 \cdot \frac{s}{c} \cdot \cos\alpha_m \cdot \left(\tan\alpha_1 - \tan\alpha_2\right) - C_{DP} \cdot \tan\alpha_m \tag{13}$$

with s/c representing the spacing to chord ratio of the blade.

Cumpsty (1989:351), offers an empirical relation for the secondary drag coefficient which includes the ratio of tip clearance to blade span t/h:

$$C_{DS} = 0.7 \cdot C_L^2 \cdot \left(\frac{t}{h}\right) \tag{14}$$

Combining these values into Equation (10) for the overall drag coefficient, the passage pressure loss coefficient can then be determined from Cohen et al.(1987:199):

$$\varpi = \frac{C_D \cdot \cos^2\alpha_1}{\cos^3\alpha_m \cdot \left(\frac{s}{c}\right)} \tag{15}$$

III. Experimental Apparatus

The work for this thesis was conducted in the AFIT Cascade Wind Tunnel Facility located in Room 143 at the Air Force Institute of Technology. The facility consists of several major components necessary to operate and acquire data. These include the air supply system, stilling tank, throat, test section, and data acquisition system. The newest modification to the facility is the moving wall upgrade to the test section.

Air Supply System

The facility, shown in Figure 3, utilizes a 30 kW centrifugal blower and high pressure air injection system which provide airflow at a rate of 1.6 kg/sec. The blower is separately enclosed in its own room and uses a combination of outside air and laboratory air as desired to regulate settling tank temperature. Input air passes through an electrostatic filter prior to the blower.

The input air next enters a 3 meter diffuser which feeds into a large settling tank where the air is further slowed to a flow velocity of approximately 3 m/s. The settling tank is constructed of a large steel pressure vessel with several layers of filter elements and honeycomb mesh to filter and straighten the flow. The settling tank is lined with sound absorbing material and has a large central plug to ensure that blower noise is not transmitted to the test section. The flow is then directed through a ASME long radius bellmouth nozzle for a smooth transition to the cascade throat. For more details of the compressor cascade facility, refer to Allison, (1982).

Data Acquisition System

The AFIT Cascade facility data acquisition system consists of four separate subsystems including a central computer, pressure monitoring system, temperature measurement system, and a probe traverse system. The data acquisition system is outlined in Figure 4.

Computer. The central computer for the data acquisition system is a Zenith Data systems Z433 D+ Computer with a 486 DX 33 MHz processor. The software was written in MS-Basic and Quickbasic 4.5. The software package includes a series of common modules shared by the acquisition and reduction programs and integrates numerous software routines provided by the individual component manufacturers.

Pressure Measurement System. The pressure measurement system consists of a Pressure Systems Inc. Model 8400 Pressure Scanner, three 32 port pressure transducers, a CEC Model 2500 Digital Barometer for monitoring ambient pressure, and a five-tube total-pressure rake for flowfield sampling.

The Model 8400 uses a vacuum pump, and a high-pressure air source regulated to 100 psi as reference sources for pressure scanner calibration. The pressure scanner has an internal certified standard transducer for use during calibration. The pressure scanner operates at a scanning rate of 20 kHz and samples each transducer port 11 times and computes an arithmetic mean of the 11 samples during each scanning cycle. The pressure transducers contain internal heaters to minimize the effects of ambient temperature fluctuations. During testing, pressure output results were periodically verified using a bank of H_2O manometers to confirm accuracy.

The transducer blocks are connected to 5 rows of static pressure ports located on the tunnel wall downstream of the cascade, nine static ports at the throat, the tunnel settling tank, and the total pressure rake. Static pressure computed for any given x location was the arithmetic mean of nine sampled static-pressure ports located on the test section root wall as shown in Figure 5. The nine static pressure ports are evenly spaced at 0.46 cm intervals and cover the primary testing passage as well as 0.8 cm into each secondary passage. Only the first row of static ports, located 0.23 chord behind the trailing edge, was used in this investigation.

The five-tube total-pressure rake shown in Figure 6 was specifically designed to examine the boundary layers along the test section walls while minimizing any blockage effects with a cross sectional area less than 3.5% of the passage. It has an outside envelope width of 1.671 cm (0.65 in.) using 5 tubes with 0.381 cm spacing from centers of adjacent tubes. This rake has tubing dimensions of 0.147 cm outside diameter, and 0.1016 cm inside diameter.

A CEC 2500 digital barometer provides an ambient pressure reference for the integrated software package and was equipped with an IEEE 488 interface to automate pressure inputs to the central computer.

Traverse Control Subsystem

The traverse control subsystem is responsible for accurate positioning of the total pressure rake in the test section. This subsystem consists of a three axis traverse aligned to the test section coordinate system shown in Figure 7. The y (pitch axis), and z (span

axis) coordinates are input and maintained by digital stepper controllers manufactured by New England Affiliated Technologies. The Model 310 Programmable Motion Controllers are individually programmable and are connected to and run by the central computer during operation with a positioning resolution of 0.0013 cm. Positioning in the x-axis is accomplished through a stepper motor and a digital encoder which accept manual inputs. The digital encoder readout allows accurate x-axis placement prior to collecting data. Pressure rake x positioning was held constant throughout the data runs.

Test Section and Moving Wall System

The test section for the AFIT compressor cascade facility houses the blades and interfaces with the moving wall system. The cascade has the specifications and design flow parameters listed in Table 1. The majority of the values listed in Table 1 are identical

Table 1. Cascade Specifications and Representative Values

Specification	Symbol	Value
Blade Span	h	5.08 cm (2 in.)
Blade Chord Length	c	3.81 cm (1.50 in.)
Blade Spacing	s	2.54 cm (1.0 in.)
Blade Camber Angle	θ	30.51 deg
Blade Inlet Angle	α_1'	26.51 deg
Blade Outlet Angle	α_2'	-4.00 deg
Diffusion Factor	DF	0.332
Point of Maximum Camber	a	0.5 chord
Stagger Angle	ζ	7.50 deg
Air Inlet Angle	α_1	31 deg
Air Outlet Angle	α_2	3.05 deg
Deviation Angle	δ	7.05 deg
Incidence Angle	i	4.49 deg
Deflection	ε	27.95 deg
Inlet Velocity	V_1	137 m/s
Outlet Velocity	V_2	107 m/s

to those used by Spacy, (1993) and Costello, (1993). Only the chord and spacing have been changed for this investigation.

The test section for the AFIT compressor cascade test facility is shown in Figure 8. The test section interior measures 22.86 cm (9 in.) between the upper and lower walls, 30.48 cm (12 in.) along the x-axis from the throat to the exit, and 5.08 cm (2 in.) wide. The cascade contains 8 cantilever mounted blades which divide the interior into 9 flow passages as well as adjustable upper and lower tailboards to balance test section static pressure. The current design does not include wall suction or throttling capability.

The basic test section is constructed of 6061-T6 aluminum with a Plexiglas sidewall cover. The blades are mounted into the root wall and enclosed by a full aluminum tip wall 5.55 cm wide that mounts between the throat and Plexiglas front cover. This arrangement is shown in Figure 5. With the installation of the rotor apparatus, the full aluminum tip wall is replaced by two machined aluminum half walls which follow the curvature of the rotor and form the tip wall above and below the region where the rotor enters the test passage. The rotor can be moved in the z-axis to change the blade tip clearance. The half walls are then adjusted up or down to provide a constant clearance gap between the rotor and contoured portion of the half walls. To limit leakage with the rotor in place, the half walls are set to provide a 0.0635 cm (0.025 in.) gap along the rotor face. Half wall gap clearance is set using a feeler gauge while the test section is at operating temperature to prevent rotor contact from thermal expansion of the tunnel.

The previous blades for this test section had a 5.08 cm chord and a 4.2 cm. They were supported from both sides using retractable pins that held each blade at two points on either side. With the introduction of a moving wall, the support arrangement was no longer workable considering that the blades must now be supported from only one side in a cantilever arrangement. Rather than attempting to modify the existing blades to be cantilevered, smaller blades were designed with a chord of 3.81 cm (1.5 in.) and a spacing of 2.54 cm (1 in.) between blades. In order to have the moving wall effects established prior to the blade leading edge, the rotor was designed to provide 0.635 cm (0.25 in.) of coverage ahead of and behind the leading and trailing edges of the blades. The rotor face width was 5.08 cm (2.0 in.) which fit comfortably in the test section with provisions for thin seals on either side of the rotor to minimize leakage. The maximum width of the rotor was limited by the width of the full aluminum tip wall that it replaced.

Once the rotor was installed into the test section, a gap in the front wall remained 0.127 cm upstream of the blades, and 0.254 cm downstream. During operation, however, the rotor exhibits a maximum wobble of 0.0762 cm (0.03 in.) out of its rotational plane which occurs at a single speed range around the natural frequency of the rotor system. This wobble requires that the spacing in the gap seals allow sufficient space to minimize contact with the rotor. On the upstream side, 2 layers of metal foil tape attached to the throat were sufficient to reduce the gap. Downstream of the blades, a Teflon scraper was inset into the Plexiglas test section front wall. This scraper nearly contacts the rotor flange allowing only enough space to permit any rotor wobble. A double layer of masking tape is placed on the leading edge of the Plexiglas front face just contacting the rotor as a

second seal. The tape is flexible enough to move with the rotor, allowing for rotor wobble, during operation while helping to minimize leakage.

A major requirement in this experiment involved approximating a flat wall with the edge of a large circular disk. As a result, a small variation in test section width is unavoidable due to the radius of curvature of the rotor. This variation was minimized by designing the rotor as large as possible for the available space. Referring to Figure 9, the maximum change in the passage width is 0.099 cm at the extreme upper and lower bounds of the testing window. The rotor line-of-symmetry is located 0.127 cm above the primary testing passage centerline. The deviation of the rotor line-of-symmetry from the primary passage centerline was caused by unevenness of the laboratory floor. Table 2 shows the wall variation due to radius of curvature for several pitch locations. Highlighted cells show locations of blades 4 and 5 with respect to y-zero reference.

The test section y-zero reference was retained from previous experiments, and therefore does not coincide exactly with the center of the testing passage. Table 2 indicates the pitch locations of blades 4 and 5. Only these two center blades model the moving wall influence. Blade 4, as counted from the top of the test section (see Fig. 5), is located at 0.55 pitch above the zero reference, and blade 5 is located at -0.45 pitch. At either blade location, the maximum variation from a flat wall due to rotor curvature is 0.0298 cm or 0.58% of the test section width.

Table 2. Variation in Moving Tip Wall Location due to Radius of Curvature

Y Pitch Location (y/pitch)	Z Wall variation (cm)
+1.00	0.09906
+0.75	0.05455
+0.55 (blade 4)	0.02981
+0.50	0.02481
+0.25	0.0066
0.00	0.000
-0.25	0.0066
-0.45 (blade 5)	0.0202
-0.50	0.02481
-0.75	0.05455
-1.00	0.09906

The rotor was positioned nearly symmetrical with the primary test passage in order to have the same tip clearance at blades 4 and 5. The approximation of the rotor as a flat wall was only reasonable for small distances along the edge, therefore, having the rotor symmetrical with the center of the passage minimizes effects due to curvature for the primary passage as well as isolating the primary passage from any leakage across the rotor/half-wall gap.

Tests were conducted with conditions matched as closely as possible between successive runs with the blade gaps set after pre-warming of both the test section and rotor. The clearance gap to chord ratios tested were 0.0033, 0.010, 0.0173, and 0.024.

Blades

The compressor blades used in this test were NACA 64-A905, a = 0.5 sections as shown in Figure 10. The blade profiles are slightly modified to smooth the flow transition from the curved part of the suction surface to the straight tail section. This blade cross section is identical to those used by Costello, (1993) and Spacy, (1993), however, the section profile dimensions have been scaled by 0.75 resulting in a new chord of 3.75 cm (1.5 in.) and aspect ratio, AR, of 1.33. The original test section used a blade chord of 5.08 cm (2 in.) with an AR of 1.0. Designing the blades with a reduced chord and pitch maintained the same spacing to chord ratio of 0.668 as in previous experiments while reducing the required rotor diameter necessary to simulate a flat wall across the passage. The blade airfoil section, spacing to chord ratio, and other characteristics are similar to previous experiments. Retaining the earlier blade profile also allowed for comparison of data and took advantage of previously coded CNC machining routines at the AFIT model shop for the fabrication of components.

Table 3 lists the coordinates for the blade cross section used in testing. The blades are constructed of 2024-T3 aluminum and were milled to shape on a Digital CNC Milling Machine at the AFIT Model Fabrication Facility. As seen in Figure 11, the 8 blades used in the cascade are mounted into the root wall of the test section on mounting blocks which are machined integral with the blades for increased strength and so as to minimize vibration. The mounting blocks insert into milled pockets in the root wall and are secured

with screws. This mounting arrangement permits the blades to be cantilevered, leaving the tips unconstrained for the study of tip clearance effects.

Table 3. Blade Surface Coordinates and Mean Camber Line

Suction Surface X (cm)	Suction Surface Y (cm)	Pressure Surface X (cm)	Pressure Surface Y (cm)	Mean Camber Y (cm)
3.751	0.000	3.751	0.000	0.000
3.677	0.015	3.675	0.007	0.011
3.527	0.046	3.524	0.021	0.033
3.303	0.089	3.373	0.035	0.062
3.193	0.111	3.184	0.052	0.081
3.008	0.152	2.994	0.069	0.110
2.709	0.214	2.693	0.099	0.156
2.410	0.269	2.392	0.127	0.198
2.108	0.315	2.093	0.149	0.232
1.803	0.342	1.798	0.160	0.251
1.498	0.345	1.503	0.158	0.251
1.308	0.336	1.318	0.151	0.244
1.118	0.320	1.133	0.142	0.231
0.929	0.297	0.947	0.129	0.213
0.740	0.267	0.761	0.113	0.190
0.551	0.228	0.575	0.092	0.160
0.362	0.179	0.388	0.065	0.122
0.287	0.157	0.313	0.053	0.105
0.212	0.131	0.238	0.040	0.086
0.138	0.101	0.162	0.026	0.063
0.065	0.063	0.086	0.010	0.037
0.029	0.040	0.047	0.002	0.021
0.012	0.025	0.026	-0.002	0.011
0.000	0.008	0.008	-0.003	0.003
0.000	0.000	0.000	0.000	0.000

IV. Experimental Procedures

Tailboard balancing

The exit pressure of the flow in the cascade test section was balanced using two screw adjusters which vary the exit area by moving the upper and lower tailboards (see Fig. 8). The static pressure on the tailboards was simultaneously monitored from four pairs of static ports located along each board. With a bank of four water manometers, each tailboard was balanced to equal atmospheric pressure at the ports. This was accomplished individually for each tailboard and then finally the matched pairs of ports were compared against each other by connecting opposing ports to a single U-tube manometer. From this comparison the tailboards balanced to within an average of 0.75 in. water, or 0.027 psig.

System Preparation

Prior to collecting data, the air system is run to preheat the test section to within the operating range of 39-44 ^{o}C. Preheating can require up to two hours depending upon room and outside air temperature. During pre-heating, the blade tip clearance changes by up to 0.0254 cm (0.010 inches).

Due to the large diameter of the aluminum rotor, temperature variations have a large effect on the non-dimensional blade gap due to thermal expansion. It is critical that the rotor reach and be maintained at a constant temperature during operation. From the first order equation for thermal expansion, thermal growth δ_T is a function of the coefficient of thermal expansion, the change in temperature, and the length of the object:

$$\delta_T = \alpha \cdot \Delta T \cdot L \tag{16}$$

where α is the coefficient of thermal expansion $23.7^{x}10^{-6}$ cm/(cm ^{o}C) for aluminum. For a rotor radius of 33.18 cm , this equates to a δ_T of approximately 0.00254 cm (0.001 in.) for each 3^{o}C of temperature change as verified by measurement. During testing, rotor temperature was held to within approximately 1^{o}C representing the largest uncertainty and hence the largest source of error in the system.

Monitoring of rotor temperature was accomplished with a self adhesive liquid crystal thermometer identical to those used on home aquariums mounted on the side of the rotor 16 cm from the hub. Its light weight was easily counterbalanced using a cardboard cutout of similar size and weight taped to the rotor opposite the thermometer. The thermometer was readable throughout the range of rotor speeds with the aid of a strobotach synchronized with the rotor rotation. The thermometer was accurate to within 0.5 ^{O}C and was unaffected by the rotor rotational speed.

The most consistent results were achieved by initially pre-warming the system for approximately one hour while using an additional heat source such as a small hand held hair dryer or heat gun to aid in pre-warming the rotor. Once the test section and settling tank were at operational temperature, the system was turned off and the blade gap set or checked as needed and the rotor temperature noted. While it is certain that thermal gradients existed in the rotor during operation, the thermometer provided a relative reference for rotor temperature assuming that the preheated tunnel temperature did not change significantly after the blade gap was set and measured. One limitation of the liquid crystal thermometer was that the highest temperature sensed was 31^{o}C, therefore, the

blade gap was set with a rotor temperature of 29.5 ^{o}C to allow for accurate measurement in the event of temperature changes during operation. Refer to Appendix B for details of rotor system adjustment. After setting the gap, the system was again turned on and the rotor temperature maintained.

The tendency of the rotor was to grow warmer as the room heated up from extended operation or when operated at high speeds. To prevent changes in the blade gap, temperature was regulated by active cooling of the rotor using cool air from the house compressed air system as well as cool outside air blown onto the rotor surface. In some cases, the rotor temperature would begin cooling below the reference value and additional heat was applied to the outer rim using the heat gun.

Blade Gap Corrections and Sources of Uncertainty

This investigation centered on measuring the effects of tip clearance and wall speed on passage and region losses. Any source of variation away from an intended gap setting presented a source of error. During the course of data collection, several factors were present resulting in possible sources of uncertainty in the tip clearance. To accurately access data as well as preclude rubbing of the rotor with the blades at close gap settings, it was necessary to determine the actual blade gap during testing.

The largest source of uncertainty stems from the variability of the tip gap between the rotor face and the blade tip due to thermal effects. Changes in temperature in the rotor result in thermal strain growth which changes the blade gap from the intended value. The tip clearance between the blades and the rotor is affected by thermal expansion during

operation due to warm-up of the tunnel to an operating temperature of 39-44 ^{0}C from a typical pre-run room temperature of 20^{0}C. Allowing for the coefficient of thermal expansion for aluminum of approximately 23.76^x10^{-6} cm/(cm ^{o}C), the rotor radius will increase by approximately 0.0177 cm (0.007 in.) and the blades by 0.00237 cm (0.000936 in.) during a warm-up temperature change of 22^{o}C. Due to the combined thermal effect of the rotor, blades, and test section, the tip clearance for the rotor and blades decreases by approximately 0.0254 cm (0.010 in). Thermal effects were minimized by monitoring the temperature of the rotor during operation, as well as cooling the rotor to maintain the temperature noted when the blade gap was set. While minimized, however, thermal effects could not be eliminated entirely.

Data taken with a stationary rotor showed losses below predicted values. With the rotor in a single position for the stationary runs, local heating of a single part of the rim led to thermal gradients and higher degrees of thermal strain than seen with the moving rotor. The asymmetric heating resulting from only one segment of the rotor exposed to the heated tunnel flow precludes assumptions of a constant thermal gradient as occurred for the spinning rotor. This uneven heating would be expected to cause a decrease in blade gap and therefore generate the lower than expected losses for zero speed cases as seen later in Figure 45.

A second source of gap change was due to centrifugal strain in the rotor due to high rotation speeds. Several methods were attempted to actively measure the rotor size during stationary and spin operation. The construction of the rotor prohibits any measurement requiring physical contact with the rotor surface.

From Timeshenko (1947) the predicted centrifugal strain was calculated as 0.0020 cm (0.0008 in.) for a uniform circular disk of the rotor diameter. Due to the flanged design, however, and for completeness of data, the actual strain growth of the rotor was measured. The rotor radial growth was measured at four speeds including 0, 1900, 2161, and finally 2620 rpm which represented the highest speed used in this experiment. A traversing microscope equipped with a digital readout and calibrated to 0.000254 cm (0.0001 in.) was used for this measurement. The microscope was clamped to a rigid stand separate from the rotor to minimize vibration. A strobotach was used to visually stop the rotor apparent motion at each speed to allow for accurate measurement. The rotor's eccentricity of 0.0038 cm was readable during the rotor growth testing; however, difficulties in exactly synchronizing the strobotach during measurements led to an uncertainty of up to 0.0015 cm at maximum rotor speed. Rotor radius changes due to centrifugal strain are presented in Table 4.

Table 4. Rotor Strain Growth Verification

ROTOR SPEED RADIANS/SEC	ROTOR RPM	Predicted Strain (Timeshenko)	Measured Strain (cm)
15.70	196.5	0.0000 cm	0.000
118.33	1480.3	0.000644	0.00101
154.98	1938.8	0.00110	0.00139
174.88	2187.7	0.001407	0.00190
209.43	2620	0.00201	0.00342

The rotor strain growth in the testing range shows some variation from the predictions based on Timeshenko's analysis, but is relatively close considering measurement uncertainty and the assumptions made in modeling the rotor as a uniform circular disk. Using these results, it was decided that effects due to centrifugal strain growth become large enough to introduce a measurable uncertainty in the tip clearance at speeds above 2000 rpm.

This growth accounts for approximately 0.0025 cm of tip clearance decrease at the highest tested speed. The result of centrifugal strain would be a decrease in gap size and therefore lower observed losses. From Equation 22 (developed later), a gap error of 0.0025 cm (0.001 in) due to centrifugal strain would result in a 0.00025 change in ϖ or an approximate 1% change in value. This was arrived at by computing the change in expected ϖ_{tip} for t/c = .010 and then the resulting ϖ_{tip} for a tip clearance 0.0025 cm (0.001 in) larger than desired.

This method of gap determination was also attempted during data runs, however, during long runs, small changes in the traversing microscope position made measurements unreliable. This was probably due to vibration transmitted through the floor due to the tunnel and the rotor systems.

While tip clearance uncertainty was the largest source of error, additional error could be introduced due to inaccuracies in the data measurement equipment. When calculated for the worst combination of component errors for an actual set of data, the largest pressure-loss coefficient error was found to be 0.5%. The published accuracy's of the instrumentation, and the uncertainties due to rotor size are presented in Table 5.

Combining the worst cases of thermal, centrifugal, and component errors, the largest uncertainty in pressure-loss coefficient was 0.0019 or 3.5%, the majority of this error due to thermal effects on gap size. The computed maximum-expected error also includes centrifugal uncertainties despite their predictable nature and applicability only to the highest wall speeds tested.

Table 5. Maximum Error Expected due to Instrumentation and Rotor Size

Source of Error	Accuracy
Pressure Transducers	0.0005 psid
Digital Barometer	0.005 psia
Thermocouples	0.3 ^{o}C
Tip Clearance centrifugal (at 2600 rpm)	-0.00254 cm
Tip Clearance - thermal	±0.00508 cm

Probe Alignment and Positioning

Positioning of the total pressure rake after initial alignment was accomplished through the Neat 310 controllers and the computer data acquisition system. All measurements were taken at an x location 23% chord (0.35 inch) downstream of the blade trailing edge. This position corresponds to the first row of root wall static pressure ports. Data runs were made using a vertical traverse with the rake aligned parallel to the blade span.

In order to evaluate losses across the test passage, the span was divided into three regions as shown in Figure 12. Each region measures 1.67 cm across and corresponds to the width of coverage of the five-tube total-pressure rake. Due to the rake width, there is overlapping coverage in the outer tubes of the center region with the inner tubes of the rake in the right and left positions. For this reason, when mass averaging data for the entire span, the center span coverage was taken as the center three tubes of the rake with the remaining area covered in the two side regions. This ensured that spanwise locations were not covered twice.

Accurate positioning of the pressure rake was critical to consistent data due to steep pressure gradients in the boundary layer. Using the Neat 310 controllers, positioning between successive runs was accurate within 0.0013 cm. Details of this procedure are covered in Appendix A.

Data Reduction

During each data run, the raw data was collected and saved in a raw data file containing:

Settling tank total pressure

Static pressure from the root wall static ports

Atmospheric pressure measured from the CEC 2500 Digital Barometer

Ambient temperature from a thermocouple in the laboratory

Total temperature from thermocouples in the settling tank

Rake total pressure for each data point.

The integrated software package contains routines for reducing raw data into a form necessary for plotting with various software packages. Settling tank total pressure was adjusted by a factor of 0.985 for total pressure losses measured by Spacy (1993) and Costello, (1993) associated with the stilling chamber and nozzle. The software contains provisions for up to 11 pressure tubes as with the original total pressure rake. For the 5 tube rake, the data was reduced by eliminating the 6 unused columns of raw data from each pressure rake data file. The data was then manipulated to present the desired output calculating either total pressure loss

$$P_{loss} = P_{tank} \cdot 0.9985 - P_{rake} \qquad (17)$$

at each data point or the computed flow velocity for each data point using the known values of P_{rake}, p_{static}, P_{tank}, and T_{tank}.

The reduction software uses the method and equations outlined in the theory section, Chapter III. The incremental area, ΔA from Equation (8) for this experiment is 0.381cm (0.15 inch) in width corresponding to the pressure rake tube spacing, and 0.0838 cm (0.033 inches) in pitch as the step size in y.

After each data run, the mass-averaged pressure-loss coefficient and mass flow are computed for each region in the same manner as the passage values were computed in Equation (9). These are recorded as well as the values of the numerator $\sum_{i,j} (\varpi_{local} \cdot \dot{m})$

and denominator $\sum_{i,j}\dot{m}_{local}$ for use in computing the passage mass-averaged pressure-loss coefficient.

The contribution of each region of the total span losses, can be computed by Equation (18). This single region value is given by:

$$\varpi_{region} = \frac{\sum\left(\varpi_{local}\cdot\dot{m}_{local}\right)_{region}}{\dot{m}_{tip}+\dot{m}_{center}+\dot{m}_{root}} \tag{18}$$

Simple addition of the values from the three regions that make up the test window result in the mass-averaged pressure-loss coefficient for the test passage as follows:

$$\varpi_{span} = \frac{\left[\varpi_{tip}+\varpi_{center}+\varpi_{root}\cdot\right]}{\dot{m}_{tip}+\dot{m}_{center}+\dot{m}_{root}} \tag{19}$$

Contour plots of pressure-loss coefficient were constructed with the pressure rake data by taking three successive data runs at root, tip, and center span which were then individually reduced. The resulting files were then combined using an in-house routine, written for this investigation, to generate a single data file for plotting using Surfer 3-Dimensional graphing software by Golden Software.

V. Results

Baseline Verification

Cascade performance was verified to establish a flowfield baseline without the moving wall modification. Initial verification used the flat, full tip-wall shown in Figure 5 which has a blade tip gap of 0.030 cm (0.012 in.) or t/c of 0.008. The 5 tube pressure rake was positioned at the center region (see Fig. 12) to verify periodicity. Figure 13 shows a survey of 101 pitchwise points taken with 0.1016 cm spacing covering a 10.16 cm (4 inch) pitchwise window encompassing the 4 central blades of the cascade. The total pressure losses were very regular and exhibited good periodicity with the blade wakes well defined and similar with respect to thickness and pitchwise spacing. This center survey did not show evidence of losses due to tip clearance effects or the wall boundary layer.

To establish the performance of the newly constructed blades, the five-tube total pressure rake was used to survey the test window shown in Figure 9 with the full tip wall. The primary test window includes the area from 2.54 cm above the cascade zero pitch reference to 2.54 cm below and encompasses both middle blades (blades 4 and 5), the primary testing passage between them, and one-half passage above and below. The center, tip, and root regions were examined to identify the locations of the corner vortices, tip vortices, and the extent of the boundary layer influence. This fixed-wall survey established a baseline for later observations as well as a means of identifying any changes caused by the moving wall modification.

The baseline contour plot shown in Figure 14, represents a linear cascade with the previously mentioned tip clearance and the full tip wall. The root wall on the right, shows a well defined corner vortex with a maximum pressure loss coefficient of 0.575 in a small region at its core. The center span is essentially free of losses except for well defined wakes associated with the two blades covered by the testing window. The tip region shows noticeably higher pressure losses and well defined tip leakage vortex cores. The majority of the losses occur in the portion of the passage closest to the suction side of the blades. (For this and all other contour plots presented here, the pressure surface is at the top of the blade shown, and the suction surface below.) The loss regions associated with blades 4 and 5, (upper wake, lower wake respectively) are very similar in shape and location. The similarity in contours of the two indicates good periodicity as well as uniformity of construction of the blades.

A second baseline was established for the cascade using half-walls with the rotor installed as shown in Figure 5. The tip clearance at blade 4 for the stationary rotor baseline was at 0.0381 cm which was 0.008 cm (0.003 in) larger than the full wall baseline. This increased tip clearance should have slightly higher losses but otherwise be very similar to the fixed wall case. Figure 15, shows the stationary wall case with the rotor installed. There is a noticeable difference from Figure 14 between the size of the tip clearance vortices and loss regions associated with blades 4 and 5. Due to the curvature of the rotor surface, although there is symmetry about the passage centerline of the primary test passage, the curvature becomes more pronounced in the secondary test passages above blade 4 and below blade 5. This increased curvature and the presence of

any associated leakage from the wall gaps appears to significantly increase the losses in the secondary test passages, a result most apparent in the tip losses below the lower blade in Figure 15.

Comparing the losses from the two baseline cases, the mass-averaged pressure-loss coefficient for the tip region of the fixed wall baseline in Figure 14 is 0.082 for both a smaller window defined by the primary passage and the full testing window. The primary passage of the stationary rotor baseline has a pressure loss coefficient of 0.0894 while the pressure loss coefficient for the entire testing window for the stationary rotor case was 0.1019. It must be emphasized, that the tip clearance for the two baseline cases differed by 0.008 cm.

The primary passage loss values for the flat wall and stationary rotor vary by 8.3%, while the pressure losses for the complete testing windows vary by 19.5% as presented in Table 6. It must be emphasized, that the tip clearance for the stationary rotor baseline is 0.008 cm larger than that of the full tip wall case. As shown later, if the stationary rotor baseline is adjusted for the larger tip gap, the differences in primary test passage losses becomes 6.1%. While the stationary rotor baseline's slightly larger tip clearance accounts for some of the increased losses in the primary passage, it does not significantly reduce the discrepancy in the full testing window, one-half pitch above and below blades 4 and 5. Based on this analysis, only the primary testing passage losses between blades 4 and 5 were used to compute pressure-loss coefficients.

Table 6. Baseline Verification Comparison of Passage Losses for Tip Regions

Verification Test	Tip clearance (cm)	Pressure loss coefficient	% from baseline
fixed wall baseline (primary passage)	0.0304	0.082	0
fixed wall baseline (full window)	0.0304	0.082	0
stationary rotor (primary passage)	0.0381	0.0894	8.2
stationary rotor (full window)	0.0381	0.1019	19.5

Analysis of Tests with Relative Wall Motion

During this investigation, data was collected at a total of ten relative wall speeds for each of four tip clearances. At each tip clearance, six contour plots of mass-averaged pressure-loss coefficient were constructed for a stationary baseline and five values of flow coefficient ϕ ranging from 20 to 1.66.

The flow coefficient ϕ is defined as the axial velocity of the flow divided by the blade tip velocity, C_x/U, and becomes undefined for the case of a stationary wall. For this investigation, relative wall speeds are therefore evaluated as the inverse of flow coefficient ϕ^{-1} (U/C_x). Values examined range from ϕ^{-1} of 0.00 for the stationary wall case, to ϕ^{-1} of 0.667 for the highest relative wall speed of 91.48 m/sec. Tip clearances are

nondimensionalized as the ratio of tip clearance over chord, or t/c. Spanwise locations are nondimensionalized as the z location divided by half the blade height, z/(h/2). Pitchwise locations are nondimensionalized by the spacing y/s. All spanwise locations are relative to the blade midspan while pitchwise locations are relative to the passage y-zero reference (see Fig. 9) unless stated otherwise.

Tip Clearance of 0.33% Chord. The smallest of the tip clearances tested was 0.0127 cm (0.005 in) for a t/c of 0.0033. Figure 16 through Figure 20 show the progression of ϕ^{-1} from 0.00 to 0.477 for this blade gap. Since the tip gap decreases slightly with increased rotor speed, the highest, safe wall speed tested for this small t/c of 0.0033 was 65.4 m/s (ϕ^{-1} = 0.477).

The stationary rotor case for this gap setting, is shown in Figure 16. The contour plot shows very compact tip clearance vortices with the high loss region smaller than that of the baseline cases (which had a larger gap). The tip vortex cores are well defined and located in nearly the same relative span location as the corner vortex at the root wall. Pitchwise, the tip leakage core is displaced slightly lower than at the root. With the exception of the somewhat higher losses due to its small tip clearance, this case is nearly a two-dimensional linear cascade as plotted by Kang and Hirsch, (1991). It is interesting to note that on the pressure surface (upper side) of the blade root for each case there is a small kink in the loss contour in the stagnation where the blade meets the wall. As seen later, any relative wall motion or larger tip clearance removes this effect at the tip. Thus, this smallest tip clearance, with t/c << 1, has pressure-loss contours very similar to a two-dimensional cascade.

The loss contours in Figure 16 show a maximum ϖ of 0.725 at the tip as opposed to 0.575 at the root. This 45% increase in loss illustrates the dramatic effect of even relatively small tip clearances on losses.

With the onset of wall motion at ϕ^{-1} of 0.0495, Figure 17, small changes in loss contours become evident. The highest contour lines have contracted slightly toward the tip wall with little reduction in core intensity or position. The location of tip losses have also begun shifting downward toward the pressure side of the adjacent blade. The tip and root region pressure surfaces near the walls are now obviously of different character, with the root region and center region, however, essentially unchanged from the baseline plots in Figure 16.

By ϕ^{-1} of 0.148, Figure 18, marked changes in contours have begun to occur. The tip clearance vortex cores are no longer well defined, but can be seen to have moved toward the moving wall region of the testing window. The tip region contours no longer resemble those at the root wall due to the shifting of the core location and pitchwise stretching of the highest loss contours.

The high loss contours in Figure 19, ϕ^{-1} of 0.28, have essentially the same spanwise location as in Figure 18 for ϕ^{-1} of 0.148, but now the lossy region is stretching in the pitchwise direction as it becomes entrained in the wall boundary layer and moves toward the pressure side of the adjacent blade. Again, the root wall contours show no influence of the moving wall.

At ϕ^{-1} of 0.477, in Figure 20, the entrainment and contraction of the vortex core is well advanced. What was previously a well defined core is now a small high-loss region

contracted toward the tip wall. The core intensity appears relatively constant compared to Figure 16, but the passage blockage due to the presence of the tip vortex has decreased considerably with the majority of the secondary effects isolated within 0.17 span of the wall.

Figure 21 shows the progression of the tip region losses and vortex location with increasing wall speed and the entrainment of the loss contours into the moving wall boundary layer. Note a courser contour interval was used in this figure to highlight the predominant effects of wall motion on the tip losses.

In summary, with the rotor stationary, tip clearance vortices remained close to the suction side of the blade (true also for all tip clearances tested) as seen in Figure 16. This behavior is typical to that described in the literature for a compressor cascade, Kang and Hirsch (1991). As the relative wall motion is initiated, however, even at relatively slow speeds, noticeable changes occur in both the pressure-loss coefficient values and the pressure contours. As ϕ^{-1} increases further, the high loss regions associated with the tip vortices became entrained by the moving wall boundary layer and were carried toward the pressure side of the adjacent blade as seen also by Kang and Hirsch (1993:448) and Lakshminarayana et al.(1995:339). This interaction also tended to narrow the high loss region toward the tip wall, reducing the overall blockage in the passage.

<u>Tip Clearance of 1.0% Chord.</u> For the gap/chord of 0.010, shown in Figures 22-27, the progression of the tip clearance vortices is much the same as for the smaller gap size just examined. The major difference at this gap size is the size and extent of the high loss region associated with the tip leakage. For the smaller t/c of 0.0033, the tip region core

intensity is 0.725, while with this larger gap setting, the core has grown substantially larger as well as more intense, with the core loss intensity in excess of 0.80. In addition, the degree of entrainment with wall speed is much more pronounced. By ϕ^{-1} of 0.28 in Figure 25, the high loss region has contracted by 0.15 in the z direction, and the core region has moved from y/s = 0.45 in Figure 22 to a y/s location of 0.35 below the suction side of blade 4. At ϕ^{-1} of 0.667 in Figure 27, the core has all but disappeared, with the pressure loss contours becoming nearly parallel to the wall. Figure 28 shows the progression of tip region losses due to the increased gap width.

<u>Tip Clearance of 1.73% Chord.</u> For an increased gap/chord of 0.017, shown in Figures 29-34, the influence of the tip clearance vortex into the passage is significantly more pronounced at the low values of ϕ^{-1} than for smaller gaps. This could be due to an increased jet flow through the tip gap. As with the earlier cases, the center and root regions of the span show no effect from either the increase in tip clearance or the effects of wall motion. In this case, as before, the core region, represented by the highest loss contour, progresses nearly halfway down the passage toward the neighboring suction side as seen in Figure 35. The high loss contour is still identifiable and never completely disappears along the wall.

<u>Tip Clearance of 2.4% Chord.</u> A tip/ chord of 0.024 was the largest gap tested in this experiment. The effects due to wall motion continue to show the same progression in losses and vortex location as in earlier cases. Seen in Figure 36 through Figure 41, the core is noticeably larger than the previous cases, extending 0.25 spanwise into the passage. At ϕ^{-1} of 0.66 in Figure 41, the core has contracted against the wall with the

highest loss region still recognizable but moved pitchwise nearly halfway down the passage toward the next blade. The progression of the tip region is shown in Figure 42.

The contour plots show that the cascade modified by increasing tip clearance maintains the losses of a linear cascade at the root and center regions. Had any changes in the root wall vorticity or losses occurred they could be attributed to the influence of either the moving wall or tip clearance. For this experiment, however, root wall losses remained essentially constant regardless of tip clearance and wall speed. . Table 7 summarizes the relative changes in tip region ϖ with increasing t/c, and ϕ^{-1}.

Table 7. Tip Region Mass Averaged Pressure Loss Coefficient for Various Gaps and Wall Speeds

ϕ^{-1}	wall speed	Tip / Chord Ratio			
U/C_x	m/s	0.0033	0.010	0.0173	0.024
0.00	0.00	0.0324	0.0323	0.0360	0.0405
0.0495	6.76	0.0320	0.0334	0.0358	0.0381
0.098	20.29	0.0312	0.0321	0.0355	0.0380
0.28	38.40	0.0297	0.0308	0.0330	0.0368
0.477	65.42	0.0276	0.0297	0.0324	0.0349
0.667	91.48	N/A	0.0298	0.0321	0.0348

Combined Effects of Tip Clearance and Wall Motion

In the contour plots of the previous section, it was apparent that the variations due to tip clearance and wall speed were pronounced and systematic for the tip region, while the center and root region losses were effectively constant for all cases. This is illustrated by the pressure loss data in Figure 43 where the pressure loss contributions of the three regions are plotted together. The pressure loss coefficients in Figure 43 were computed as shown in Equation (18). While some small variations occur in the root region and center region, losses are markedly independent of tip clearance (in general agreement with Lakshminarayana et al., 1986:28) and changes in ϕ^{-1}. Compared to the substantially higher losses of the root and tip regions, the center region's contribution is negligible for the tested blade. Since the distribution of mass flow in the passage is unaffected by relative wall motion and tip clearance, the argument can be made that the tip region can be examined independent of the center and root regions of the passage.

To determine if leakage through the half-wall rotor gap was affecting loss values, a set of data was collected using small cardboard flaps which rested on the rotor edge as an additional seal against leakage through the halfwall rotor gap. This set of data, for t/c = 0.017, showed slightly higher, (approximately 2%), losses than that without the flaps. This difference, however, fell within the scatter due to other sources of uncertainty and therefore did not alter the correlation with the other data. This method for minimizing leakage was not repeated due to evidence of wear on the rotor face following the test. A method could possibly be devised to further reduce leakage without causing rotor wear,

however, the result should only be a slight increase in loss data but similar overall trends in terms of tip clearance and wall motion.

Mass Flow Distribution

While the behavior of pressure losses in the tip region are strongly dependent upon wall speed and tip clearance, the mass flow distribution is essentially immune to these effects. The mass flow distribution follows the inverse of the region losses with the highest mass flow associated with the center region which has the lowest loss coefficient. The mass flux distribution is given in Table 8 using an average mass flows from Figure 44. As seen in Figure 44, the root and center region mass flows are essentially constant regardless of tip clearance or wall motion. The mass flux for each region was computed by dividing the regions average mass flow by the fraction of span contained in that region. The low losses and hence limited blockage to flow in the center of the span make its relative mass flux greater. Overall, the mass flux distribution between the smallest and largest tip clearances vary by less than 2%, with the variation even less when based upon wall speed.

In this investigation, the tip and root regions were determined by the width of the total pressure rake used to collect data. Ideally, the tip region would extend only to the limit of three-dimensional tip or root influences. In Figure 44a, the mass flow for the tip region normalized by the passage mass flow is plotted for each Δz position. The area under the curves represents the total tip region mass flow. The mass flow distribution shows the effect of losses with those spanwise locations with the highest losses having

correspondingly lower mass flows. Despite the redistribution with pressure losses, Figure 44 shows that the mass flow for the region is essentially constant. The z location 1.5 cm into the passage is the apparent limit of pressure losses on mass flow distribution. Beyond this point, the mass flow has reached center region values.

Noting that mass flow distribution is apparently linked to loss coefficient could possibly lead to a correlation between loss data and region mass flow. While a correlation may exist, it should be noted that despite the large disparity in losses between the three regions shown in Figure 43, the mass flow fractions of the regions are much less sensitive.

Table 8. Average Mass Flux Distribution by Region

	Tip Region	Center Region	Root Region
Fraction of Span in region	0.378	0.225	0.378
t/c	Mass Flux Distribution (Region Mass Flow /Region Fraction)		
0.0033	0.957	1.093	1.032
0.010	0.942	1.097	1.037
0.024	0.934	1.102	1.039

<u>Analysis of Three-Dimensional Effects Using Tip Region Pressure Losses</u>

The results of the region pressure-loss coefficients, mass flow distributions, and the contour plots support the assertion that moving wall and tip clearance effects do not play an important role in the center span and root regions. The distribution of $\overline{\omega}$ versus

gap size (see Eqn. 18) in Figure 45, shows a high degree of consistency in the data trends. With this in mind, an empirical formula was found which correlates the data for the series of gap clearances into a single curve. Using a single set of data as a baseline, the remaining gap lines can be collapsed onto the baseline using the tip/chord ratio of the separate data series:

$$\varpi = \varpi_{ref} \cdot \left[1 + 11.35 \cdot \left(\frac{t}{c} - \frac{t}{c_{ref}} \right) \right] \qquad (20)$$

The reference loss, ϖ_{ref}, represents the mass-averaged pressure-loss coefficient for the reference gap size, with t/c_{ref}, the tip to chord ratio, corresponding to the reference data. Using this relation, the data from the four tested tip clearances were collapsed onto the reference data as shown in Figure 46. Error bars are included in Figure 46 for each value of U/C_x. These bars represent the standard deviation of the data from its mean at that U/C_x.

From the collapsed data, an empirical equation was derived which approximates the reference data line. Using an exponential fit with the variables optimized by the application of a least squares technique (Holman, 1978:64)., the following expression was developed.

$$\varpi_{ref} = \left[(0.0340 - 0.0298) \cdot e^{\frac{-U}{0.20 \cdot C_x}} + 0.0298 \right] \qquad (21)$$

Combining Equations (20) and (21), the mass averaged pressure loss coefficient for any combination of relative wall speed or tip clearance within the range tested is given in Equation (22).

$$\varpi = \left[(0.0340 - 0.0298) \cdot e^{\frac{-U}{0.20 \cdot C_x}} + 0.0298 \right] \cdot \left[1 + 11.35 \cdot \left(\frac{t}{c} - 0.010 \right) \right] \qquad (22)$$

It is important to remember that the empirical relationship is only valid for the blade section tested for the given test conditions and incidence angle (fixed loading). It is possible that this technique could be extended to include variations in chord and incidence with further testing.

An immediate application of this relation is to account for the difference in tip clearance from the fixed wall baseline to that of the stationary rotor baseline. The measured difference in losses was 0.082 versus 0.0894 for t/c = 0.010 (rotor), an 8.2% difference. The stationary rotor losses for t/c = 0.008 results in a tip region loss of 0.0873 for a difference of 6.1% for the simulation.

The correlation developed for tip region data was mass averaged by the total mass flow of the passage. This mass averaging tends to minimize the differences in tip region losses. Figure 47 shows the tip region losses mass averaged by the tip region mass flow. While this means of presenting the data does not permit the simple addition of region values to find the passage losses, it is illustrative concerning the magnitude of changes attributable to tip clearance.

The tip region size in this investigation was based upon the width of the total pressure rake. The tip region, however, is best defined by the depth of penetration of tip losses into the passage. If the tip region size was deemed that region which encompasses the major losses, tip clearance and relative wall motion effects could be compared separately from the remainder of the passage. This suggests a possible means of analyzing and applying this data in turbomachinery. Dividing the passage into regions of dominant influence would allow losses for a new passage to be estimated given only the chord, span, and loading. An example of this is presented in Appendix C.

Analysis of a Crenulated Blade Design

Having developed and verified the capability of the moving wall modification, a set of data was generated for a possible a follow-on application of this facility, the study of crenulated blades. Several investigations were previously conducted on modifications of stator and rotor blades with notched crenulations in the trailing edge DeCook,(1993), Spacy (1993). The purpose of these crenulations is to improve wake mixing and perhaps decrease overall passage losses.

Three blades were modified with a crenulation design studied by Spacy (1993) and termed long-narrow crenulations. This design uses four crenulations 0.635 cm wide and 0.9525 cm deep cut into the trailing edge as shown in Figure 48. The blades studied by Spacy (1993) had a chord of 5.0 cm, while the current blades have a 3.81 cm chord. To accommodate the present blades, the crenulation design was scaled in chord by 0.75 while the span dimensions and crenulation locations remained unchanged.

The crenulated blades were tested with the moving wall rotor in place over the full range of wall speeds at a tip clearance of 1.46% of chord (this t/c was chosen because repeatable data was obtained with similar clearances for straight blades). The contour plots generated from these data runs are shown in Figure 49 to Figure 54.

As seen in Figures 49-54, the behavior of the crenulated blades is very similar to the straight blades with respect to wall motion and the entrainment of the tip region losses into the moving wall boundary layer. As seen in Figure 44, the mass flow distribution for the crenulated blades was essentially identical to that of the straight blades.

The crenulations appeared to influence the behavior of the tip and root wall vortices. A comparison of, say, Figure 29 ($t/c = 0.0173$, $U/C_x = 0$) with Figure 49 shows that the root wall vortex interacted with the crenulation closest to the wall. The high loss region associated with the root vortex became separated into two distinct segments in the regions of $2z/h = 0.5$ and 0.8, the former located at the near-root wall crenulation. The core region of the root wall vortex was pushed toward the root wall and the region of highest intensity appeared reduced in size. The changes in the core size and intensity, however, appeared to be countered by a small increase in the overall extent of the corner loss region. The overall root wall losses (see Figure 43a) compared closely with values collected on the straight blades.

Some variations in root wall losses occurred, but appeared to be due to overall experimental scatter rather than any established trend. For most points, the root (non-moving) wall mass flow values were identical (Figure 44) to those of the straight blades.

The center span also exhibited the effects of the crenulations. As seen in Figures 49-54, the blade wake in the center region widened (in comparison with straight blades) implying enhanced downstream mixing at the tested x location 0.23 chord behind the trailing edge. The losses in the center region were marginally lower than those seen typically with the straight blades, but the overall differences were minor.

The tip region losses for the crenulated blades, shown in Figure 56, matched the exponential decay for the straight blades and interestingly were near the predicted pressure-loss coefficient values from the empirical equation for a clearance of t/c = 0.0146.

Comparison to Empirical Data

Cohen et al. (1987), separated losses into those due to secondary effects, annulus losses, and profile losses. Further, an expression for the passage drag coefficient C_D is simply the summation of its various components, and the pressure loss coefficient for each source can be calculated by a slight modification to Equation 15. For the secondary losses alone, C_D is replaced by C_{DS} using Equation 14.

$$\varpi = \frac{C_{DS} \cdot \cos^2 \alpha_1}{\cos^3 \alpha_m \cdot \left(\frac{s}{c}\right)} \qquad (23)$$

Table 9 shows the contributions of the various parts of the pressure loss coefficient based on the empirical relations of the theory section and the values found in this investigation.

To determine an empirical prediction of the root region losses, the components can be summed in a manner such that the root region losses include about half of the secondary losses, an equal share of the annulus losses with the tip region, and a portion of profile losses matching its percentage of the span. The predicted value for the root will be slightly high considering that secondary losses also include tip effects. From this comparison, the root losses calculated from the empirical equations of Cohen et al. (1987) are at worst 15% low as shown in Table 9, and for the higher tip clearance values, very close indeed. In equation form this becomes:

$$\varpi_{root} = \frac{(\varpi_{DS} + \varpi_{DA})}{2} + 0.378 \cdot \varpi_{DP} \tag{24}$$

The predicted tip losses, ϖ_{tip} can be computed in a similar manner. The tip predictions are consistently lower than experimental data which suggest the empirical relations do not adequately account for variation in the tip region.

The total loss predictions underestimate the actual data by 11% to 14% with the agreement improving for larger tip clearances. The largest error is due to the calculation of secondary losses based upon the tip clearance. If the 6.1% correction for the moving wall simulation is applied to the data for tip losses, the total passage loss coefficients agree within 7% for the 2.4% chord tip clearance.

Table 9.   Predicted vs. Experimental Region and Passage Pressure Loss for Large U/C_x

tip/chord	0.0033	0.010	0.0173	0.024
C_{DS} (Eqn 14)	0.000735	0.00275	0.00478	0.00662
C_{DA} (Eqn 11)	0.01	0.01	0.01	0.01
C_{DP} (Dixon,1986:78)	0.022	0.022	0.022	0.022
C_D (Eqn 10)	0.03273	0.03475	0.03678	0.03862
ϖ_{DA} (Eqn 23)	0.0128	0.0128	0.0128	0.0128
ϖ_{DP} (Eqn 23)	0.0282	0.0282	0.0282	0.0282
ϖ_{DS} (Eqn 23)	0.0009317	0.003486	0.00605	0.00849
ϖ_{root} predicted (Eqn 24)	0.0177	0.0189	0.0202	0.2149
ϖ_{tip} predicted (Eqn 24)	0.0177	0.0189	0.0202	0.2149
ϖ_{total} predicted (Eqn 15)	0.0420	0.00446	0.0472	0.04957
ϖ_{root} data (Figure 43)	0.021	0.021	0.021	0.021
ϖ_{tip} data (Figure 45)	0.00276	0.0298	0.0323	0.0346
ϖ_{total} data ($\varpi_{tip}+\varpi_{center}+\varpi_{root}$)	0.0486	0.0508	0.0533	0.0556

Pressure Rise Coefficient, Cp

Despite the importance of losses, one must remember that this investigation concerns modeling a compressor stage and that the purpose of a compressor is to generate a pressure rise in the flow. The pressure rise coefficient Cp is defined as the increase in static pressure across the stage normalized by the dynamic pressure.

$$Cp = \frac{p_2 - p_1}{\frac{1}{2} \cdot \rho \cdot V_1^2} \qquad (25)$$

The Pressure Rise Coefficient Cp, plotted in Figure 57 shows some sensitivity to variations in tip clearances and wall speed. For the straight blade cases, Cp is generally higher for the smaller tip clearances, however, this is only consistently apparent for t/c = 0.024. For t/c = 0.024 Cp values ranged 5% to 10% below that of the smaller tip clearances. Some decrease would be expected with increasing blade gap considering a larger tip clearance would lead to increased leakage and lower static pressure rise for the cascade. Cp for the crenulated blade was very close to that of the straight blades except at the stationary rotor case where it was nearly 12% higher than for the straight blade cases. While interesting, this higher Cp at the root would need to be confirmed by additional testing before attempting to draw a conclusion. Cp values were taken from the center region with the rake at the y-zero reference position.

VI. Conclusions and Recommendations for Further Research

Conclusions

A modification of the AFIT Compressor Cascade Facility was designed, constructed, and tested to permit the study of three-dimensional effects of a moving sidewall in a linear compressor cascade. The moving annulus wall was simulated using a large aluminum rotor. The following conclusions can be drawn from this investigation into the effects of a relative wall motion and tip clearance on the tip clearance vortex and losses in a blade passage:

1. For the stationary rotor case, mass-averaged pressure-loss coefficients agreed to within 6.1 % (rotor losses larger) of those with a solid wall (with 0.008% gap). Pressure loss contour plots closely resembled those of a solid wall for the primary test passage, indicating only a minor effect of rotor curvature on losses.

2. Operation of the moving wall at various tip clearances and wall speeds allowed measurement of a smooth progression from stationary cascade flow structures to those with a moving annulus wall. Tip leakage vortices which were well developed and located near the suction side of the blade in all stationary wall cases became entrained by the moving wall boundary layer. These vortices degenerated into narrow regions of pressure loss located progressively further along the tip wall with increasing wall speed.

3. Analysis of pressure-loss contours, mass flow distributions, and mass averaged pressure loss coefficients support the conclusion that losses associated with tip clearance and wall motion are confined to a region in the vicinity of the tip wall which is consistent

with observations seen in rotating compressor rigs. (e.g., Lakshminarayana et al., 1986:30). Tip region losses exhibited a continuous decrease with increasing wall speed and appeared to follow an exponential decline to a final steady state value. An empirical relation was found which relates tip region losses to a function of tip clearance and wall speed for the particular incidence tested. The derived relationship correlated the four sets of tip clearance data onto a single curve.

4. A crenulated blade design was tested for losses with tip clearance and relative wall motion. Passage losses were in close agreement with both comparable straight blades and the empirical correlation, while wake mixing appeared enhanced by the crenulated design.

Recommendations for Further Research

The following are recommendations for further research based upon the findings of this experiment.

1. Although this investigation reached a minimum flow coefficient of 1.66, the installation of a more powerful motor and a means of throttling axial throughflow would increase the capability for flow coefficients in the region of 0.5. Lower flow coefficients would further validate the endwall loss correlation and extend its range of applicability.

2. The results in this study were obtained with lightly loaded blades at a single blade incidence angle. Higher incidence throats already manufactured for the AFIT compressor cascade could be matched with a modified headstock which would permit the study of losses in highly loaded blades with relative wall motion and tip clearance. This

could potentially generalize the current empirical relation to include blade incidence as well as confirm its applicability for the full range of compressor operation. High incidence angles up to stall would represent the most demanding case of lift coefficient and tip region losses.

3. Flow field measurements did not include effects of gap and rotation on flow turning (deflection). Future measurements should include flow angles and velocities in various regions downstream of the cascade.

4. The current setup could also be modified to include endcaps on the blades to simulate shrouded blades.

5. The effects of high inlet turbulence was not studied. The inlet section could be modified to include air injection to study this effect.

6. The effects of rotation and gap on stalled flow, though not studied in this project, could be simulated by blocking the flow into the test passage. By blocking successive passages in turn, a steady-state flow field simulation of the unsteady stall event could be studied.

7. The boundary layer attached to the rotor was not examined. Future studies should include velocity measurements of the incoming velocity profile to determine the extent of skewness due to rotation and whether repositioning of the rotor to create a longer wall region adjacent to the blade leading edge would be required. (this could be done by including a spacer between the tank nozzle and the test section or by machining new blades or repositioning them 0.25 in further downstream.

8. Current crenulated blade designs have been optimized to enhance mixing of the vortices present in a linear cascade. With the entrainment of tip clearance vortices by the

annulus boundary layer, tip region losses no longer resemble those in a two-dimensional cascade. With this in mind, crenulation designs could be optimized for three-dimensional flow effects similar to those seen in turbomachinery.

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Appendix A. Pressure Rake Alignment

Data is strongly dependent upon consistent placement of the pressure rake in the z or spanwise direction. Due to steep gradients in the wall boundary layers, z position errors can result in substantial variations in data. To aid in consistency, all z traverse measurements were referenced off the fixed root wall using feeler gauges to position the outer wall of the nearest tube of the pressure rake at 0.127 cm (0.05 inches) from the root wall as shown in Figure 12. The rake was then traversed to the tip wall again using 0.127 cm clearance. The center region places the midline of the center tube 2.54 cm from the root wall. Once the rake was positioned, all subsequent traversing was done using the Neat 310 controllers. With the stepper motor controllers calibrated in inches, the right and left window positions were reached by traversing the probe 0.605 inches either side of the center location. At the conclusion of each series of data runs, the probe was traversed back to the center location in order that this reference would not be lost after the controllers were turned off. Positioning was periodically cross-checked against scribe lines on the traverse mount.

All measurements in this study were taken from an x location of 23% chord (0.35 inch) downstream of the blade trailing edge. This position corresponds to the first row of root wall static pressure ports. The x stepper motor and controller allow consistent placement of the total pressure rake in the test section. While not computer controlled, an x traverse controller and readout allows accurate manual positioning of probes into the test section. For alignment, the total pressure rake is first installed in the probe holder and

locked into place using the locking screw. The rake was then advanced into the test section and was squared against the root wall using a combination square. The rake was then aligned with the trailing edge of the test section and the numeric readout from the controller noted. The rake was advanced into the test section with the Plexiglas side wall removed until the probe was in line with the static pressure ports corresponding to the desired testing position. The numeric readout is again noted and becomes the x reference for subsequent runs. As long as the probe is not removed from the holder, these readings will prove accurate. If it is necessary to remove the probe and reinstall, the same relative numeric difference on the controller readout will give the same x location providing similar alignment procedures are performed. Finally, the rake was positioned immediately behind the blade trailing edge to check that the rake tubes were parallel to the blade.

Y positioning of the pressure rake is referenced from a set of scribe lines on the Plexiglas side wall. The scribe lines correspond to approximately the middle of the primary test passage and are used to provide a constant y reference. Due to slight shifting of the rake with air flow on, the rake is positioned in y by traversing the probe to the left sampling window, and with the air flow turned on, the midpoint of the nearest tube is placed in line with the opposing scribe lines on the Plexiglas side wall. The positioning is done with the stepper motor program through the Neat 310 controllers. Once the y position is adjusted, the rake is traversed back to center passage, and the new reference point is saved. Provided the rake is repositioned back to center prior to removing power, the reference position will not need to be reset.

Appendix B. Moving Wall Design and Construction

A major part of this investigation involved the conceptualization, design, and construction of the modifications required to conduct moving endwall studies on the AFIT compressor cascade facility. The resulting modifications included the rotor assembly, contoured front walls, and cantilever blade mounting system.

The rotor assembly consists of 4 major parts as shown in Figure 58:

pedestal assembly

headstock

motor and mount

rotor assembly

Pedestal

The pedestal is constructed in two sections to ease portability and maximize stiffness and adjustability, with the largest lower section weighing over 100 lb. The pedestal contains two sections of 4 inch by 6 inch I beam which are joined by two mating plates of 1/2 inch steel plate. Due to flexure problems noted during initial rotor spin testing, an additional 3 inch flange was welded to the inner web of each I beam section to increase the overall stiffness about the shorter axis of the I beam.

The lower base section is welded to a 3/4 inch steel plate base measuring 19.5 inches square. A series of large gussets reinforce the lower I beam base plate joint. The entire unit is bolted to the floor in operation to increase rigidity. Because final positioning is critical, the pedestal is designed with approximately 1 inch of adjustment in each

direction with adjustment slots at each set of mating plates. In addition, adjustment slots in the headstock base where it joins the pedestal allows the rotor to be moved aside for testing with the solid faceplate when moving wall effects are not desired.

Headstock

The headstock consists of a machined 1/2 inch steel plate welded at a 59 degree angle aligning the rotor spindle with the test section flow and positioning the rotor edge in the test section. The spindle, hub and bearings are taken from a 1985 Dodge Omni rear axle spindle, hub and bearing assembly. Off the shelf automotive quality bearings were chosen due to their designed operating loads similar to our anticipated values. Typical wheel rotational speeds for a small automobile at 90 mph with an 18 inch diameter wheel is in excess of 1680 rpm.. Tapered roller bearings used in automotive applications, however, are designed for a 50,000 + mile life, therefore, running at twice this speed but much shorter duration should be reasonable considering that Timken roller bearings are generally rated to over 3500 rpm in other applications. The simple design of these assemblies permits the rotor disk and hub to be dismounted from the spindle by the removal of a single nut and cotter key prior to removing it as a unit.. Despite limited working space in the laboratory, when installed, adjustments can be reached allowing for modifications of test conditions with respect to blade tip clearance.

To change the blade tip clearance gap, loosen the headstock bolts so that they will allow sideways movement but still support the rotor and act as alignment guides. For this operation, a ratchet with an 18 inch extension and swiveling socket are required. The

front half walls should be loosened and backed off prior to adjusting the rotor position. To decrease gap clearance, place a feeler gauge of the desired thickness along the center of the blade and gently move the rotor until the gauge moves with only a mild resistance. Light taps with a leather mallet can aid in making the final adjustments. Forcing the rotor rapidly may result in bent blades. Once the gap is set, retighten the headstock bolts ensuring beforehand that the motor drive pulley has clearance from the back of the rotor or scraping may occur when the rotor turns. Next, the half wall gap should be set. Ensure that there is at least 0.038 inch (0.028 in. on a warm system) between the half wall contours and the rotor face, or contact may occur during operation. Finally, replace the Plexiglas side wall and use a double layer of masking tape to make a final seal between the Plexiglas side wall and the rotor.

For periodic maintenance and greasing the bearings, the rotor can be removed from the headstock by first backing it away from the test section and removing the belt from the rotor pulley. Next, remove the cotter key , bearing nut, and washer. A slight outward pull on the rotor will pop out the front bearing. Remove the front bearing and then the rotor can be lifted off. Have a sturdy table cleared and close at hand prior to lifting off the rotor, it is awkward and weighs nearly 80 lbs. To reinstall the rotor just reverse the procedure. The spindle nut should be tightened and the cotter key reinstalled. Once the rotor is reinstalled, the drive belt can be adjusted using the screw shaft tensioner or a suitable pry bar to place tension on the belt. If the belt is placed at its closest setting to the blades and tensioned initially, it should not have to readjusted for the small clearance changes encountered during testing.

Motor and Mount

The motor mount consists of a 3x3 angle iron support welded to a half inch steel plate motor rest with both pieces bolted to the upper pedestal I beam section and oriented parallel to the spindle. The motor is raised on four collars with interior guide washers to help raise the motor relative to the headstock and increase the range of motor adjustment. The drive belt is tensioned before operation by loosening the four motor hold-down bolts, using the screw tensioner to tighten the belt allowing minimal slack,(ensure that the motor pulley remains parallel to the rotor face during tightening) and then retightening the motor hold-down bolts. The motor mount is also adjustable for spacing between the rotor and pulley. If the motor mount is removed for any reason, the mount should be readjusted to align the pulley grooves so that the belt will run smoothly.

The installed motor is a 1 HP DC Baldor industrial motor rated for 2500 rpm. It is operated by a DC Controller with variable speed from 0-2500. An RPM readout is included which operates from an electronic pickup mounted to the motor face. The tachometer gets its reference pulse from the 60 tooth gear on the motor shaft. As the teeth pass the pickup, it generates a square wave output that is read by the speed control card in the Controller. The Controller uses the tachometer output as feedback in maintaining constant motor RPM. The motor RPM can be read from the LCD display connected to the Controller. The final pulley drive ratio of 1 to 1.31 from motor to rotor was verified using a strobotach tachometer. A simple table was constructed for laboratory use to determine rotor RPM as a function of motor speed.

Rotor Assembly

To achieve the desired face speed of 300 fps as well as maximize the rotor radius of curvature, the largest rotor was designed that would fit within the available laboratory space. Considering that internal stress in the rotor increases with the square of the rotation speed as well as the problem of bearing loads and motor requirements, the rotor was designed for an upper limit of 92 m/s (300 fps) for the rotor edge speed. This corresponds to 2630 rpm or 43.84 rps for the final rotor diameter of 66.37 cm (26.133 in.). A rotor edge speed of 92 m/s was desired based upon average turbomachine with an 12 inch diameter normally operates in the range of 5000 to 20,000 rpm. At 5,000 rpm or 83.3 rps, the tip speed of the rotor blades relative to the wall is approximately 260 fps. The design goal was to reach a wall to blade tip relative speed as close as possible to an actual engine. The combination of stresses and lab space resulted in a final rotor diameter of 66.37 cm.

With numerous possible choices of materials for the rotor disk, aluminum was chosen for its known material and thermal properties, machinability, availability and relatively light weight. The rotor was machined from a solid plate of 7075-T7 aircraft aluminum which has a yield stress of over 60,000 psi. A drawback of this particular alloy, is its tendency for stress cracking. It is highly recommended that prior to the start of an investigation, that the rotor be removed and checked for any crack growth in the region

near the hub attachment bolts. This area is the most highly stressed and has the added stress concentrations due to the bolt holes.

Modeling the rotor as a solid disk of 2.54 cm width but with an 3.81 cm of diameter to account for the additional material of the flange, the maximum stress calculates to 4800 psi at 3000 rpm and 6850 psi at 3600 rpm using the methods and equations outlined in (Timeshenko, 1947). A flange equal to the rotor web width increased the edge thickness from 2.54 cm at the web to 5.08 cm at the edge. The flanged design was used in order to reduce the overall weight of the rotor disk while still maintaining confidence in operational stress values. For the design speed, a safety factor of over 10 times allows for any inherent material flaws.

The large aluminum rotor is securely mounted on the pedestal which tilts the rotor to the angle of the blade cascade which is determined by the desired turning in the test section. For this experiment, the turning is 31 degrees, the same value used in 2 previous thesis as well as representing a diffusion factor of about 0.33 to remain clear of any complexities associated with possible stall in highly loaded blades.

. From the attached schematics Figure 59, the rotor assembly is bolted from the back side through the driven pulley with four precision ground spacing washers between the pulley and disk to provide clearance for the drive belt. The hub is inserted full length through the rotor and ends in a flange which sandwiches the rotor between it and the pulley. The hub assembly is held by the four castellated nuts on the hub side. The bearings are contained in the hub with the larger rear bearing held in place by the rear bearing retainer and remains with the hub except when removed for greasing. The front

bearing must be removed whenever the rotor is dismounted from the headstock. It is held in place by a keyed washer and finally a nut, nut retainer, and cotter key. A grease cap protects the bearings from dirt.

While aluminum has many good properties resulting in its selection, its relative softness requires that great care be exercised to prevent nicks and scratches during use.

Further Refinements

In the course of testing, several possible refinements were considered that would improve the accuracy and ease of use of the moving wall modification. While not incorporated during construction due to lack of time, a screw feed placed on the headstock where it joins the upper pedestal mating plate would greatly ease the accurate resetting of blade gaps. Since the only area where accuracy is required is in closing the blade gap, a screw fixed to the headstock which pushes against the pedestal would allow accurate headstock positioning while using the headstock attachment bolts as guides.

Perhaps the largest problem associated with the rotor mechanism is that of ensuring constant tip clearance during operation. This was handled indirectly during this investigation through temperature control of the rotor. The optimum solution would be to measure and adjust the blade gap nonintrusively during operation Knowing the gap is only part of the problem, controlling it is also required. A possible solution to this problem might involve enclosing the rotor in a transparent housing with a small room air conditioner providing cool air to the housing during operation. The worst heat buildup occurs during high speed testing. At high RPM the rotor bearings begin to warm and the

rotor does not have time to dissipate the heat it picks up while crossing the test section. Ideally, an automatic thermostat triggered by the temperature of the rotor would work however this presents the problem of remotely sensing the rotor temperature while in operation. A much simpler way would involve using the liquid crystal thermometers that are currently in use and controlling the temperature manually with controls for temperature located near the rotor assembly.

While adequate for this investigation, the 1 HP Baldor motor proved only marginally capable of providing the power needed for timely acceleration and maintenance of maximum RPM during testing. A 1.5 to 2 HP motor of at least 2500 rpm maximum speed with the same mount dimensions could be purchased and incorporated on the existing mount. This coupled with a means of throttling the air flow could potentially permit reaching values of flow coefficient very close to those seen in turbomachinery.

Appendix C   Scaling Data for Application of Tip Wall Correlation to Different Spans

The purpose of developing a tip wall correlation is to permit application of experimental loss data to blades with similar characteristics but different spans. The data collected for the three blade regions can be scaled appropriately to any given blade of similar airfoil section, loading, spacing and chord.

The data in this investigation was for a blade with a 5.08 cm span. In order to apply this data to a blade of larger span the pressure loss regions must be adjusted accordingly. For the new blade, the width of the tip and root regions should not change. The center region, however, will need to expand to cover the larger center region of the larger blade. This is based upon the assumption that the loss regions at the tip and root are fixed by the blade characteristics and loading, with the tip clearance, moving wall effects, and annulus losses remaining restricted to the tip and root regions.

In this investigation, the tip and root regions were determined by the width of the total pressure rake used to collect data. Ideally, the tip region would extend only to the limit of three-dimensional tip or root influences. In Figures 60 and Figure 61, the tip region loss penetration is plotted against tip clearance and relative wall motion. The extent of influence in all cases was approximately 1.5 cm into the passage. For this example, however, the tip and root regions will remain as plotted in this investigation.

The widths of the tip and root regions are constant for the larger span being considered, however the percentage of span that they represent changes If the total mass

flow is normalized to 1.0, then the mass flow fractions represent the mass flows of the various regions for the experimental data. To adjust these to a new span, it is only necessary to determine the new mass flow based upon the change in width of the center region. The new center region mass flow will become:

$$\dot{m}_{center(new)} = \dot{m}_{center(data)} \left(\frac{w_{center(new)}}{w_{center(data)}} \right) \tag{26}$$

where $w_{center(data)}$ is the width of the center region for the data case, and $w_{center(new)}$ is the new center region width. $W_{center(new)}$ can be calculated by:

$$w_{center(new)} = h_{new} - w_{tip(data)} - w_{root(data)} \tag{27}$$

where h is the blade height. The new mass flow for the passage will become:

$$\dot{m}_{total(new)} = \dot{m}_{tip(data)} + \dot{m}_{center(new)} + \dot{m}_{root(data)} \tag{28}$$

The mass averaged tip and root region losses for the new case must be scaled for the new span. Figure 47 plots the pressure loss coefficient mass averaged by the tip region mass flow. Knowing the mass flow through the tip region, the scaled mass flow can be determined by knowing the ratio of tip region mass flow to the total mass flow for the data and the ratio of spans.

$$\varpi_{tip(new)} = \varpi_{tip-region(data)} \cdot \left(\frac{\dot{m}_{tip}}{\dot{m}_{total(data)}} \right) \left(\frac{h_{data}}{h_{new}} \right) \qquad (29)$$

For the root region, the mass averaged pressure loss coefficient was constant regardless of tip clearance or wall speed, therefore the root region losses for the data case are 0.0200 as shown in Figure 43 mass averaged by the total passage mass flow. This root value should resemble that of a stator due to the lack of any moving wall effects. For a rotor, the root losses would show effects due to the skewing of the flow vector at the hub.

The losses for the new span become:

$$\varpi_{span(new)} = \frac{\left[\varpi_{tip} \cdot \dot{m}_{tip(data)} + \varpi_{center} \cdot \dot{m}_{center(new)} + \varpi_{root} \cdot \dot{m}_{root(data)} \right]}{\dot{m}_{total(new)}} \qquad (30)$$

For a new span of 7.5 cm, tip region losses can be computed from Equation (21) using the correlation of tip/chord and wall speed. This example assumes the wall speed to be beyond the exponential decay point. If the new tip/chord is 0.20 for this example, the losses for the tip region for the data case is 0.0331.

The results of Equation (26) through Equation (30) for the new span of 7.5 cm are shown in Table 10.

Table 10. Scaled Passage Losses based on Experimental Data

	Data			New		
Blade span	5.08			7.5		
Region	Tip	Center	Root	Tip	Center	Root
region width	1.95	1.18	1.95	1.95	3.6	1.95
% of span represented	38.3	23.2	38.3	26	48	26
relative mass flow	0.362	0.246	0.39	0.362	0.750	0.39
total mass flow	0.998			1.5		
ω_{region}	0.0331	0.00	0.020	0.0220	0.00	0.0133
ω_{total}	0.0531			0.0353		

Appendix D. Equipment Listing

Subsystem	Model Number/Description
Pressure Measurement	Pressure Systems Inc. Model 8400 Pressure Scanner (20 kHz maximum sample rate) Model 8415 Scanner Interface Unit Model 8420 Scanner Digital Unit Model 8440 Analog Input Unit Model 8433 Pressure Calibration Unit (1 psid) Model 8433 Pressure Calibration Unit (5 psid) Certified Standard Transducer (0.2-18 psi range, 0.0001 psi accuracy)
	Transducer block P/N 32RG-0301 (1 psid range, 0.0005 psi accuracy)
	Transducer block P/N 3201B (1 psid range, 0.0005 psi accuracy
	Transducer block P/N 3205B (5 psid range, 0.0005 psi accuracy
	Pressure Rake (5 ports spaced 0.381 cm) (0.158 cm OD, 0.1016 cm ID)
	CEC Model 2500 Digital Barometer (13.00 psia - 15.51 psia range, 0.005 accuracy)
Temperature Measurement	Hewlett-Packard Model 3455A Digital Voltmeter (0.0002 V accuracy, 24 Hz maximum sample rate)
	Hewlett-Packard Model 3495 Scanner
	Omega T-type Thermocouples (2) (copper-constantan, 0.3^{o}C accuracy)

Traverse Control	New England Affiliated Technologies Model 310 Programmable Motion Controllers (2)
	Oriental Motor Company Stepper Motors (2) (400 steps/revolution resolution)
Central Computer	Zenith Model Z433D+ (80486 DX 33 processor) National Instruments Model GPIB-PCII General Purpose Interface Board) TSI Model 6260 Parallel Interface Board
	Software (developed by AFIT/ENY, written in MS-Quickbasic 4.5)
Rotor System	Baldor CDP-3450 Motor (1HP DC permanent magnet, 2500 RPM)
	KB Electronics DC Controller KBPC-240D